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(54) Title: ADJUSTMENT MECHANISM FOR VALVES (57) Abstract <p>An apparatus for adjusting the motion characteristics of a valve of an engine, comprising an adjustment means to vary the valve opening and closing angle, or the valve lift including a guide element movable from a first position to a second position, and having a path adapted to receive a guide member of a valve actuation means, the guide member received in the guide path having a trajectory, when the adjustment means is in the first position, different of the trajectory of the guide member when the adjustment means is in the second position.</p> <div data-bbox="979 1163 1401 1913" data-label="Image"> </div>		

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ADJUSTMENT MECHANISM FOR VALVES

FIELD

The present invention relates to improvements in engines, such as internal combustion engines, particularly to the actuation of valves and most particularly, poppet valves for internal combustion engines.

The present invention also has application to engines or pumps which uses valves.

BACKGROUND

The available torque from an internal combustion engine is largely dependant on the volumetric efficiency of the engine. For reciprocating piston engines, this efficiency is a measure of the volume of atmospheric air drawn into the cylinder/s during an induction stroke, relative to the swept volume of the cylinder/s.

The valve timing of the reciprocating internal combustion engine has a significant effect on the volumetric efficiency of the engine at particular engine speeds. An engine having fixed valve timing, ie a fixed crankshaft angle for valve opening before piston at top dead centre and a fixed crankshaft angle for valve closing beyond top dead centre, will have a particular engine speed where it operates most efficiently. At this speed, the fixed synchronisation of the inlet and exhaust valves opening and closing relative to the piston position create the combination giving the most torque.

Obviously it is desirable to have the most torque possible available over a wide range of engine speeds. To achieve the maximum torque at high engine speeds, it is desirable to have the valves (inlet and exhaust) open for as much piston travel as possible. This gives air more time to enter and exhaust gases more time to exit the cylinder and therefore increases volumetric efficiency. However, there are limiting factors for how much piston travel, or how much of an angle (of crankshaft rotation) the inlet and exhaust valves can be kept open. For example, increasing the angle that the valves are open increases the angle that both the inlet and exhaust valves are open at the same time, which is called valve overlap. Valve overlap is desirable at high engine speeds as it increases torque output. However, the same amount of valve overlap that produces good

torque at high engine speeds will cause the engine to run poorly and reduces torque output at low engine speeds. Accordingly, in general, opening the valves earlier and closing them later improves volumetric efficiency at high engine speed at the expense of torque at low engine speed. Conversely decreasing
5 valve overlap increases engine torque at low engine speeds but does not give the best efficiency at high engine speeds.

It is therefore desirable to have a mechanism whereby the timing of the valve opening and closing can be adjusted according to parameters such as engine speed, in order to optimise the torque across a range of engine speeds.
10 Further, other parameters, such as for example, throttle position and which gear is engaged, may be used to vary the timing of the opening and closing of the valves.

Apart from valve timing, there are other factors which are important in the operation of reciprocating engine poppet valves. Firstly, just before the valve is
15 opened, the valve actuator should accelerate slowly towards the valve, in order to reduce then eliminate the clearance between the valve and the actuator or between any intervening tappet arrangement and the actuator. This is to ensure that the valve and actuator do not impact on each other with large velocities or forces. The valve then needs to be opened as quickly as possible in order to
20 facilitate the filling of the cylinder with fresh air and fuel in the case of an intake valve, or empty the cylinder of exhaust gas in the case of an exhaust valve. Once opened, the valve should be held open for as long as possible before closing rapidly. The valve should then reseat as gently as possible and then stay closed until the cycle repeats. As there should be no radical changes in
25 motion, (excessive acceleration) of the valve, a substantially sinusoidal motion has been found to be acceptable in providing a path for valve movement.

The actuation of valves and the control of their motion has been accomplished in the past by the use of camshafts. Camshafts have an eccentric cam lobe that actuates a valve, wherein the profile of the cam lobe determines
30 the motion characteristic of the valve. A problem with this arrangement is that the camshafts spin rapidly and the valves rely on valve springs to keep them in contact with the outer surface of the cam lobe. As the camshafts spin more

rapidly, the valves can leave the surface of the cam lobe due to inertia. This problem has been addressed in part by increasing the strength of the valve spring, however this makes opening the valve harder and increases wear on the cam lobe surface.

5 Another major problem with camshafts is the inability to change the lobe shape, making modification of the motion characteristics of the valve difficult. To modify the valve timing, the camshaft needs to be replaced or machined, with the result that torque is only optimised over a narrow speed range for a particular cam lobe profile. This is one of the reasons that engines that perform
10 well at high speed usually lack torque in the lower range of engine speeds. Further, as the valve springs push against the cam lobes as the camshaft rotates, significant twisting forces are generated along the camshaft, which can result in camshaft breakage.

There are existing devices that attempt to solve some of the above
15 problems, however, none are completely satisfactory. One device is a cam shaft having two standard cam lobes for the two inlet valves, and a third cam lobe between the two standard inlet lobes. When the engine is spinning below a certain engine speed, the inlet valves are actuated by the standard cam lobes. When the engine accelerates over a predetermined engine speed, a pin
20 engages with the valve's actuators, which allows both the valves to be actuated by the third cam lobe, which has a different profile suited to high engine speeds, wherein the inlet valves open earlier and stay open longer. A similar mechanism operates in the exhaust valve camshaft. This system has the disadvantage that it is not possible to vary the valve opening and closing times
25 between the two predetermined valve motion characteristics, ie there are only two valve opening durations available. This results in a marked "step" in torque output from lower rpm to higher rpm and fails to achieve the maximum torque output across the whole range of engine speeds, as effectively only two specific engine speeds are optimised.

30 Another method of varying the valve opening and closing angle is where the camshaft speed is not always half the crankshaft speed over parts of a single revolution, but varies according to the engine speed. For example, at low rpm

the camshaft may spin at the standard rate of half the crankshaft speed. At higher engine speeds, a mechanism mounted on the camshaft causes the camshaft to spin at lower than half crank speed while opening the valves and keeping them open, thus ensuring that the valves are open over a wider angle than at lower speed. In order to make up lost time (as the crankshaft must average one revolution for every two crankshaft revolutions), the camshaft must then spin faster than half crank speed for the remainder of the revolution to ensure that it is in the correct position when it is time for the valves to open again. This system is obviously less than ideal as a complex mechanism is used to vary the speed of the camshaft with respect to the crankshaft over a single revolution. Further, valve lift cannot be modified as the cam lobe profiles cannot be modified.

Another disadvantage of most valve actuation means is that they comprise a cam shaft which opens the valves. Camshafts are difficult to manufacture, and are subject to wear and breakage.

It has also been found that the method of adjusting valve clearances between the top of the valve and the valve actuator, for example rocker arm or cam shaft lobe, has disadvantages, such as the need for the clearance adjusting mechanism to be on the rocker arm, thereby adding inertia to the rocker arm, or the use of shims which are difficult to get at under the cam lobes, and require buckets to locate them, which add to the overall length of the valve assembly and therefore add to the dimensions of the engine.

SUMMARY OF INVENTION

It is an object of the present invention to alleviate at least one disadvantage associated with the prior art.

To this end, the present invention provides a means for adjusting the motion characteristics of a valve. The motion characteristics of the valve include timing, such as the crankshaft's angular location before the top dead centre reference angle where the valve opens, duration, such as the angle of crankshaft rotation for which the valve will stay open, lift or travel the amount of lift of the valve for a given crankshaft angular location, rate of travel and/or force. In one form, the adjustment is actuated mechanically. In another form, the

adjustment means is located between a valve actuation means and the valve. Advantageously, adjusting the motion characteristics of the valve by way of the present invention, enables selection of engine performance criteria from a range of predetermined characteristics, together with a selection of the degree to which the criteria is to be performed. For example, the adjustment of valve motion characteristics may be selected in a manner which accentuates engine torque. Or, selection may be made to accentuate engine fuel economy.

It may also be desirable to produce a valve actuation means which produces an approximate sinusoidal motion of valve lift in relation to crankshaft rotation and/or which also allows the motion characteristics of the valve to be varied.

Usually the valve actuation means includes a rotating member.

Typically the adjustment means varies the valve opening angle, and/or the valve closing angle and/or the valve lift, either individually or collectively. It has been found that it is advantageous to vary the valve lift and duration, and that while these may be done separately, it has been found that it is beneficial to increase valve lift and valve spring duration as engine speed rises.

Accordingly, it is desirable that the adjustment means varies the valve opening and closing angle and the valve lift collectively.

In another form, the invention provides an apparatus for adjusting the motion characteristics of a valve, including adjustment means to adjust the valve motion in accordance with the adjustment means travel along a non-straight path.

In another form, the invention provides an adjustment means for use in an apparatus for adjusting a motion characteristics of a valve comprising a plate having a guide path.

In another form, the invention provides an apparatus for adjusting the motion characteristics of a valve including a first guide path and a second guide path wherein the motion characteristics of the valve are determined by differences in shape and/or alignment between the first and second guide paths.

In another form, the invention provides an apparatus for adjusting the clearance of a valve actuated by a desmodromic valve actuation means

including a valve having a threaded end portion.

In a preferred embodiment, the means for adjusting the motion characteristics of the valve include an adjustment member having a guide path and a pivotally mounted valve actuation member having at least one guide surface, wherein a pin moves along both the guide path and the guide surface, causing the pivotally mounted actuation member to pivot and move the valve.

Typically, the pin is driven in a substantially cyclic motion.

Desirably the guide path of the adjustment member and the guide surface of the valve actuation member are not collateral over their entire length, ie there is a difference in the paths such that they deviate from each other at least over part of their length. This difference in paths produces the movement of the actuation member as the pin travels along both paths. also, a kinematic inversion of pin and guide is contemplated as an alternative embodiment.

PREFERRED EMBODIMENT

One or more of the preferred embodiments of the present invention will now be described, with reference to the accompanying drawings, wherein:

Figures 1a-1d show a schematic representation of an adjustment mechanism in accordance with the present invention in various states of assembly;

Figure 2a shows a schematic side view of the adjustment mechanism of the present invention and a prior art valve actuation mechanism;

Figure 2b shows a schematic view of a non-desmodromic adjustment mechanism of the present invention and a prior art valve actuation mechanism.

Figure 3 shows an isometric view of part of a first embodiment of the adjustment mechanism of the present invention;

Figure 4 shows an isometric view of all of the first embodiment of the adjustment mechanism of the present invention;

Figures 5a and 5b show end views of a second embodiment of the adjustment mechanism of the present invention;

Figures 6a and 6b show end views of the adjustment mechanism shown in figures 5a and 5b;

Figure 7 is a graph of typical extremes of variation in lift and duration of

the valves compared with the position of the crankshaft, as varied by the adjustment mechanism of the present invention;

Figure 8 is a first embodiment of a guide plate of the adjustment mechanism of the present invention;

5 Figure 9 is a second embodiment of the guide plate of the adjustment mechanism of the present invention;

Figures 10a-10d are embodiments of rocker arms of adjustment mechanism of the present invention;

10 Figure 11 is a schematic side view of a first embodiment of a slot of a guide plate of the adjustment mechanism in accordance with the present invention;

Figure 12 is a schematic side view of a profiled surface of a guide plate of the adjustment mechanism in accordance with the present invention;

15 Figure 13 is a schematic side view of a second embodiment of the slot of the guide plate of the adjustment mechanism of the present invention;

Figure 14 is a schematic representation of the slot of the guide plate of the adjustment mechanism of the present invention;

Figures 15a-15d are embodiments of a sliding pin of the adjustment mechanism of the present invention;

20 Figures 16a-16d are embodiments of the guide plates of the adjustment mechanism of the present invention;

Figure 17a is a first embodiment of a guide plate adjustment means of the adjustment mechanism of the present invention;

25 Figure 17b is a second embodiment of the guide plate adjustment means of the adjustment mechanism of the present invention;

Figure 18a is a perspective view of a valve clearance adjustment mechanism in accordance with the present invention.

Figure 18b is an exploded perspective view of the valve clearance adjustment mechanism shown in figure 18a.

30 Referring to Figures 2a, 2b, 4, 5a and 5b, a mechanism 10 is shown for adjusting the motion characteristics of a poppet valve 1. The mechanism 10 includes a valve actuation means, for example a valve crankshaft 12 having a

crank pin 13, which is used to provide the cyclic displacement motion and base timing for actuation of the valve 1. The valve crankshaft 12 is normally driven by the crankshaft (not shown) by known means such as a belt or chain drive, or gears, at half the crankshaft rotation speed. The mechanism 10 is typically
5 mounted in the head of a reciprocating four stroke engine (not shown) and further includes a pivot point 14 which is fixed to the head of the engine to pivotally locate a rocker arm 16 which actuates the motion of the valve 1. An adjustment member such as a guide plate 18 is mounted to the head such that it is able to be moved within a range of positions, for example a first position 20
10 and a second position 22 as shown in figures 1b, 5a, 5b, 6a, 6b and 9.

Crank pin 13 is attached to a conrod 24 through aperture 15 at one end and has a sliding pin 26 at the other end as shown in figures 1a-1d, 2 and 3. As the valve crankshaft 12 turns, the conrod 24 moves the sliding pin 26 along a path 25 in the guide plate 18. The guide plate 18 does not move in response to
15 movement of the sliding pin 26, and the sliding pin 26 is constrained to move along path 25. The sliding pin 26 also travels along a path 28 on rocker arm 16. There is typically one rocker arm 16 per valve 1, and accordingly there may be two rocker arms 16 if two inlet (or exhaust) valves are used per cylinder. The two rocker arms 16 and guide plates 18 can be served by a single conrod 24
20 and sliding pin 26 as shown in figure 4, thus actuating two valves (inlet or exhaust) simultaneously as in a four valve per cylinder engine head. Obviously the number of valves that can be actuated by a single conrod and/or sliding pin is not limited to two per inlet / exhaust. This path 28 may be in the form of a slot having an upper and lower profiled surface, in the case of desmodromic valve
25 actuation, as shown in figures 1a, 1b, 2a, 4, 5a, 5b, 6a, 6b, 10a, 10c, 13 and 14 or it may be a single profiled surface in the case of conventional valve actuation with a spring providing the valve closing force as shown in figures 2b 10b and 10d.

In either method of valve actuation described above, the path 25 in the
30 guide plate 18 causes the sliding pin 26 to move in a way constrained by the profile of the path 25, which causes the rocker arm 16 to pivot about pivot point 14. The rocker arm 16, when pivoting, has a forked actuator arm 32 distal from

the pivot point 14 which pushes on the valve 1 via two nuts tightened on the valve stem, as shown in figures 5, 18a and 18b and described later, thus causing valve 1 to open and close according to the differences in the profile of path 25 and path 28. It is the difference in path profiles as shown in figures 13 and 14 that causes the rocker arm 16 to pivot, and the profiles of the paths 28 and 25 may be varied according to the motion characteristics desired from the valve, which may vary from engine to engine, or with the purpose of the engine. Thus the individual path profiles of the guide plate 18 and rocker arm 16 are not intended to be limited to the various embodiments shown in this specification.

10 Further, the valve 1 can be replaced by valve system 100 as shown in figures 2a and 2b, wherein valve system 100 includes a known shim and bucket arrangement that allows the valve clearance to be adjusted, and a valve spring 101 ensures that the valve stays in contact with the actuator 32 when the valve is closing, which may be used in non-desmodromic or conventional valve
15 actuation. The mechanism 10 may simply replace camshaft 102 as a valve actuation means.

The assembled parts of the mechanism 10 can be seen in figures 1a to 1d, wherein the assembly of the conrod 24 to the crankshaft 12 by the crank pin 13, and the attachment of the guide plate 18 and rocker arm 16 to the sliding
20 pin 26 is shown schematically. A partially assembled adjustment mechanism is shown in figure 3, wherein the assembled valve crankshaft 12, conrod 24 and sliding pin 26 are seen in relationship to the pivot point 14 which is normally in a fixed position, but can be rotated and includes in this embodiment eccentric section 11. The assembled mechanism 10 is shown in fig 4 wherein two guide
25 plates 18 are slidably attached to the eccentric section 11 of the rocker shaft 14 and two rocker arms 16 are pivotally attached to the pivot point 14, which would enable the mechanism 10 to operate two inlet or exhaust valves. The guide plates, in this case, are constrained to slide linearly by pins (not shown) which fit into guideways 9.

30 In operation, the valve crankshaft 12 rotates at half engine crankshaft speed. The conrod 24 is connected at one end to a crank pin 13 on the valve crankshaft 12 and at the other end to the sliding pin 26. The pin 26 is located in

a guide path 25 of the guide plate 18. As the valve crankshaft 12 rotates, the pin 26 is constrained to move along the path 25 of the guide plate 18, however the guide plate 18 can move from a first position 20 to a second position 22, and any number of positions therebetween. The profile of the guide path 25, as
5 shown in the figures, defines the trajectory of pin 26. The pin 26 also slides along path 28 of the rocker arm 16, and the different profile between the path 28 and path 25 causes the rocker arm 16 to pivot back and forth about pivot point 14. The actuator 32 attached to rocker arm 16 moves with the arm 16 and contacts the end of valve 1, pushing the valve open and pulling the valve
10 closed. Where non-desmodromic valve actuation is desired, a valve spring may close the valve 1.

The position of the guide plate 18 can be varied, in the case of Figures 2a, 2b, 3 and 4 by rotation of the rocker shaft 14, in a second embodiment adjustment 1s due to eccentric adjusting shaft 30 as can be seen in figures 5a,
15 5b, 6a, 6b 17a and 17b. The shaft 30 has an eccentric off-centre lobe 31 which can be turned within aperture 34, thus causing the guide plate 18 to move from a first position 20 wherein the motion characteristics as shown by line 40a and 40b of figure 7 suit low engine speed, to a second position 22, wherein the motion characteristics of the valve suit high engine speeds shown by line 42a
20 and 42b, also of in figure 7. The movement of the guide plate 18 can be seen in the comparison of open valve positions shown in figures 5a and 5b. In figure 5a, the adjusting shaft 30 and lobe 31 position the guide plate 18 in the first position 20. In figure 5b, the adjusting shaft 30 and lobe 31 position the guide plate 18 in the second position 22 and thus the maximum valve opening, as
25 seen in figure 5b is greater than the maximum valve position seen in figure 5a. The operation of the shaft 30 and lobe 31 in the aperture 34 in the guide plate is shown in figures 17a and 17b and will be described in more detail below.

The motion characteristics of the valves can be seen in figure 7, wherein line 40a represents the valve lift of an inlet valve (vertical axis) versus crankshaft
30 rotation angle (horizontal axis) for the valve 1 actuated by valve crankshaft 12 while the guide plate 18 is in the first position 20. The exhaust valve motion characteristics when the guide plate 18 is in the first position 20 are shown by

line 40b. Line 42a represents the valve motion characteristics when the guide of an inlet valve when the guide plate 18 is in the second position 22. The exhaust valve motion characteristics when the guide plate 18 is in the second position 22, can be seen in line 42b. As can be seen from figure 7, there is a significant difference between valve lift, valve opening duration and valve overlap when the guide plate 18 moves from a first position 20 to a second position 22.

The reason for the difference in motion characteristics is that when guide plate 18 is in the first position 20, the profile of path 28 is positioned such that the differences in the profile between path 25 and path 28 are minimised, as can be seen in figures 5a, 5b and 8 and discussed below. This provides a lower valve lift as the pin 26 deflects less, as shown in the position of pin 26a in figure 8.

The first position 20 of guide plate 18 opens the valve the least amount, and over the shortest angle, and is therefore normally used for low engine speeds where excessive valve overlap is undesirable and increased turbulence is desirable. The second position 22 of the guide plate 18 is used to generate larger valve overlap and higher lift in the valves, as seen in the position of the pin 26b in figure 8, and the extension of the valve in figure 5b compared to the extension of the valve in figure 5a. This arrangement is used during high engine speeds where maximum gas flow is required. Figures 6a and 6b show the guide plate 18 in the first position 20 and second position 22 respectively, but the valves in both cases are closed fully, i.e. regardless of the position of guide plate 18, the valves still close effectively as shown by the equal positions of the valves in figures 6a and 6b. The difference in the positions of the plate 18 is clearly seen by the gap between pivot point 14 and guide plate 18 in figure 6b, whereas in figure 6a there is no gap. The mechanism 10 allows the guide plate 18 to be positioned at any point between the first position 20 and the second position 22, thus allowing the amount of valve overlap and/or the angle of valve opening to be adjusted to any point within the predetermined limits of the valve motion characteristics. This also provides the advantage of being able to modify the valve opening angle and/or lift according to any change in the

conditions in order to maximise volumetric efficiency.

- The differences between the profiles of path 25 in guide plate 18 and the path 28 in rocker arm 16 are designed to impart the desired valve motion characteristics to the valve. For example, when used with a rocker arm 16 having a straight path 28, the path 25 shown in figure 11, is made up of four portions, each with a specific function. Portion A is a portion whereby, when the sliding pin 26 is in this portion, the valve will be closed. As the pin 26 travels along the path 25, it moves to portion B, which is a ramp section designed to allow the pin 26 to begin to move at an angle to the direction of motion in portion A. This allows the actuator 32 to be brought into contact with the valve (or valve shim) relatively slowly, as there is usually a small gap between the top of the valve assembly and the valve actuator. Once the valve has contacted the top of the valve, sliding pin 26 enters portion C of path 25, where the slope of the path increases greatly and thus causes the actuator to push open the valve quickly. Once the maximum valve opening is approached the sliding pin 26 enters portion D whereby the velocity of the valve while opening is reduced, and the valve starts to decelerate. In portion D, the sliding pin 26 reaches the end of its travel and the valve crankshaft 12 begins to pull the sliding pin 26 back along the portion D in the reverse direction, thus starting to close the valve again.
- Figure 12 shows the portions A-D of a profiled surface of a guide plate which uses a spring to return the valve to the closed position, and therefore does not require the lower portion of the path. The shape of the paths 50 and 51 in figures 11 and 12 respectively are designed to be used with a rocker arm having a substantially straight path 28.
- If the path of the rocker arm had a shape the same as the shape of the path of the guide plate, then the rocker arm would not move relative to the guide plate and accordingly there would be no motion of the valve. Therefore there are numerous shapes that either the path of the guide plate, or the path of the rocker arm can take in order to produce the required motion of the valve providing that the other of the rocker arm or guide plate has a profile that is different. As an example, the shape of the path 52 of the guide plate shown in figure 13 can be used, provided the shape of the rocker arm path 53 differs in

the correct areas to provide the motion in the rocker arm. This difference in path shapes is shown in figure 14 wherein the paths 25 and 28, have been overlapped in order to highlight the differences in the profiles which then cause the rocker arm to deflect and actuate the valve.

5 From figures 13 and 14, the differences in the paths in the guide plates and rocker arms can be seen, and the differences relate to valve lift. Figure 13 relates to path 52 in a guide plate that is adjusted rotatably, for example as shown in figure 9. It can be seen that this arrangement allows a far greater difference between paths 52 and 53, and accordingly, a far higher valve lift is
10 achieved than in the linearly adjustable guide plate shown in figure 14. This increased valve lift shown in figure 13 is accomplished without a radical increase in path deviation, which would be necessary in a linearly adjustable guide plate, such as that shown in figure 8. It is undesirable to have too large a deviation in any of the paths as this may lead to increased wear on the path
15 surfaces which will cause the valve motion characteristics to change.

An advantage of the present system is that by altering the differences in the profiles of the paths 25 and 28, it is possible to produce a valve motion with, for example, a more square top than that shown in figure 7.

In order to overcome or reduce wear due to high contact pressure
20 between the path 28 and the sliding pin 26, it has been found that the sliding pin can be made with a non-circular cross section, called a wear portion, in the region where it travels along the path 28.

In figure 15a, a wear portion 60 is shown having a flat upper and lower surface where the pin contacts a straight path 28. The profile of the surfaces
25 varies to match the facing surfaces of the paths. For example, the wear surfaces can be flat as in wear surface 60 when used with path 128 in rocker arm 116 as shown in figure 10a. Alternatively the wear surfaces can be curved with a common centre of curvature as shown by wear surfaces 360 in figure 15c to suit a similarly curved path 328 in figure 10c of constant radius. If conventional or
30 non-desmodromic valve actuation mechanisms are used, then the sliding pin only needs one wear surface 260 or 460 as shown in figures 15c and 15d, as the valve spring will ensure continuous contact of the wear surface with the

opposing path 28 surface.

It should be noted that the embodiments shown in the figures 10a-10d, 15a-15d, and 16a-16d, work together in respective sets. A rocker arm 116 having a path 128 as shown in figure 10a is used with a gudgeon pin 126 shown in figure 15a and a guide plate 118 having a path 125 shown in figure 16a. This arrangement forms an adjustment mechanism employing desmodromic valve actuation wherein the guide plate 118 is adjusted linearly.

Similarly, a rocker arm 216 having a path 228 (figure 10b) works with a gudgeon pin 226 (figure 15b) and a guide plate 218 having a path 225 (figure 16b) to form an adjustment mechanism employing a valve to close the valve, wherein the guide plate 218 is adjusted linearly.

A rocker arm 316 having a curved path 328 (figure 10c) works with a gudgeon pin 326 (figure 15c) and a guide plate 318 having a path 325 (figure 16c) to form an adjustment mechanism employing desmodromic valve actuation, wherein the guide plate 318 is adjusted pivotally.

A rocker arm 416 having a curved path 428 (figure 10d) works with a gudgeon pin 426 (figure 15d) and a guide plate 418 having a path 425 (figure 16d) to form an adjustment mechanism employing a valve to close the valve, wherein the guide plate 418 is adjusted pivotally.

In figure 17a embodiments of a mechanism for adjusting the position of the guide plate 18 is shown. The adjusting shaft 30 is situated in the aperture 34 in the guide plate 18. By rotating the adjusting shaft 30, the eccentric cam lobe 31 on the adjusting shaft 30 causes the guide plate 18 to move linearly, for example, as shown in figure 8. The amount of linear movement of the guide plate 18 is determined by the amount of rotation of the shaft 30. This allows the guide plate 18 to be adjusted to any point between and including the two extreme positions, being the first position 20 and the second position 22.

In figure 17b, the shaft 30 is rotatably received in to an aperture 134 in a guide plate 618 mounted so as to be pivotally adjustable about point 135. As the shaft is rotated, eccentric lobes 31 force the guide plate 618 to move. The guide plate is constrained to move pivotally and therefore, twisting the shaft 30 causes the guide plate 618 to move. As above, the amount of movement of the

guide plate 618 can be controlled by the rotation of the shaft 30.

A control means (not shown) is used to control the rotation of the shaft 30 for each mechanism 10 which enables the guide plate to be positioned anywhere between the first position 20 and the second position 22. The control means may be a simple device for advancing the valve opening by twisting the shaft, or any other suitable means for moving the guide plate. Such mechanisms are commonly used to advance the ignition timing as engine speed rises. The valve timing in this case may be adjusted either with or independent of the ignition timing.

10 A further embodiment of a guide plate 518 is shown in figure 9 wherein the guide plate 518 is mounted to a rotatable pivot point 535, so that adjustment of the motion characteristics of the valve can be made by rotating pivot point 535 to which the guide plate 518 is attached, to any position between the two positions as shown by the arrow and dotted line, rather than linear motion as
15 shown by the arrow in figure 8.

It should also be understood that the guide plates in any of the embodiments disclosed may be positioned in discrete locations between the first position 20 and the second position 22, for example by the use of a stepper motor. This would allow the position of the guide plates to be varied in steps
20 according to data from various parameters such as engine speed, rate of change of engine speed, throttle position and gear position. Accordingly, a fuzzy logic table could be set up to position the guide plates in the optimum position for a set of predefined parameters.

Figures 16a to 16d show further alternative arrangements for the guide
25 plates. Each guide plate is arranged to be mounted in such a way that its position is able to be controlled in order for the position of the path for the sliding pin to be controlled. In a non-desmodromic arrangement as shown in figure 16b and 16d, there is no requirement for the path to be a slot, and as such profiles 125 and 325 can be used, as a spring acting on the valve can be used in a
30 conventional manner to close the valve and accordingly there will always be pressure on either profile 125 or profile 325 and the underside of the respective rocker arms 118 or 328. This arrangement has the advantage that there is a

large body of knowledge regarding the use of valve springs to close a valve. Also, the reciprocating rocker arms may be made lighter.

Embodiments of the means for adjusting the position of the guide plate is shown in figure 17a and 17b. Figure 17a relates to a method of
5 producing linear adjustment in the guide plate using a shaft 30 in an aperture 34 in the guide plate 18. The shaft has a lobe 31 which moves the guide plate to the desired position when the shaft 30 is turned. Aperture 34 is designed to move the guide plate 18 linearly, and therefore has substantially straight side walls. As many engines of the type that use poppet valves have numerous
10 valves in alignment, a single shaft with multiple lobes 31 can be used to move all the guide plates 18 simultaneously.

A further embodiment is shown in figure 17b, wherein the shaft 30 is used to cause a rotational motion in the guide plate 18. The twisting of the shaft 30 with eccentric lobe 31 in aperture 134 causes the guide plate to pivot about
15 fixed point 135. If the guide plate is mounted about a pivot point, as shown in figure 9, then the rotation of the shaft 30 will cause the guide plate to rotate, and thus increase or decrease the difference between the paths in the guide plate and rocker arm, which will effect the motion characteristics of the valve. As the aperture 134 is designed to move the guide plate 18 pivotally, side wall 136 is
20 longer than side wall 137.

In the above embodiments, the rocker arm has pivoted while the guide plate has moved either linearly or pivotally. It can be readily determined that the rocker arm could also move linearly in response to the movement of the pin in the path of the guide plate. Further, the guide plate may be fixed in place, and
25 all the adjustment movements can take place on the rocker arm, eg the rocker arm could have its pivot point moveable with respect to the guide plate. This arrangement has the advantage that the guide plate is then fixed, and all the movement is undertaken by the rocker arm, making the mounting of the guide plate greatly simplified.

It can be seen from the embodiments disclosed that the movement of the guide plate 18 from its first position to the second position causes the sliding pin 26 travelling along path 25 to not only increase the crank rotation angle across

which the valves open, but also increases valve lift at the same time. These aspects in combination produce a result that is very desirable, as two of the valve characteristics change with only a change in one parameter, that being the movement of the guide plate. It is desirable to have the valves increase their lift at high engine speeds to ensure that the maximum amount of air enters the cylinder or exhaust gas exits from the cylinder in the time provided. However, at low engine speeds, it has been found that increased turbulence in the air entering the cylinder is desirable as it assists in the atomisation of the fuel in the air. When engines operate at low speeds, the velocity of the air entering the cylinder is also low, and therefore there is not as much turbulence in the air as it passes the inlet valves into the cylinder. It has been found that decreasing the valve lift and duration increases turbulence and therefore increases fuel atomisation, which increases torque. At higher engine speeds, the turbulence from the faster air flow provides sufficient energy for fuel atomisation, and the limiting factor becomes the amount of air able to be squeezed into the cylinder. The present invention allows for the adjustment of not only the valve opening duration, but also valve lift with only one parameter being adjusted.

The motion characteristics of the valve may also be varied in accordance with factors such as throttle position and also which gear is selected.

It should be noted that it is not essential to increase valve lift and duration with engine speed, and that it may be desirable under certain circumstances to decrease valve lift and/or durations of the inlet and/or exhaust valve as engine speed increases which the present invention is also able to accommodate.

In figure 18a there is shown a guide plate 418 used in desmodromic valve actuation, having two branches 420, each branch having an actuator 32. The actuators 32 sit between an upper flange member 422 and a lower flange member 424 at the upper end of a valve 1. The valve 1 includes a threaded portion 426, which has a lower nut 425 including the lower flange member 424 threadedly attached thereto, as shown in figure 18b. An upper nut 428, which is threadedly attached to the threaded portion 426 of the valve 1, includes the upper flange member 422. The gap between the upper flange member 422 and the lower flange member 424 may be set by an intermediate shim member (not

shown) which would fit between the upper nut 428 and lower nut 425, whereby the size of the shim determines the gap between the upper flange member 422 and lower flange member 424.

In the embodiment shown, the upper flange member 422 includes a spacer 423 which contacts a corresponding spacer 427 on the lower flange member 424, thus providing the appropriate gap between the flanges. Typically, the size of the gap is slightly larger than the diameter of the portions of the actuator 32 that contact the upper and lower flanges, thereby allowing a clearance between the flanges and the actuators 32. The upper and/or lower nuts may be held in position by lock nuts (not shown). The valve clearance may then be adjusted by removing the upper lock nut (if provided), removing the upper nut 428 having upper flange member 422 and spacer 423, and replacing the upper flange member 422 with spacer with another flange member and spacer of suitable size, then reattaching the lock nut onto the threaded portion of the valve 1. In this way, the valve clearance can be adjusted to take into account any wear in the system, without having to replace the guide plate 418. The spacer 423 may be integral with or separate to the upper flange member 422, and the lower flange member 424 may also be replaced if desired.

Alternatively, the upper nut 428 and upper flange member 422 may be locked into position by the upper nut 428, and the gap between the upper and lower flanges can be set by the position in which the upper nut 428 and flange member 422 are set.

It is important that the contact surfaces of actuator 32 be of constant radius, as in this way the valve clearance will be constant as the valve is opened and closed, and the rocker arm 16 moves about pivot point 14.

THE CLAIMS DEFINING THE INVENTION ARE AS FOLLOWS:

1. An apparatus for adjusting the motion characteristics of a valve.
2. The apparatus of claim 1 including an adjustment means operatively intermediate a valve actuation means and the valve.
3. The apparatus of claim 1 or 2, wherein the adjustment means varies the valve opening angle and the valve closing angle.
4. The apparatus of claim 1 or 2, wherein the adjustment means varies the valve lift.
5. The apparatus of anyone of claims 2 to 4, wherein the valve actuation means includes a rotating member producing a reciprocating motion.
6. The apparatus of claim 5, wherein the rotating member includes a crankshaft.
7. The apparatus of any of the preceding claims wherein the adjustment means is mechanical.
8. The apparatus of any of the preceding claims, wherein the valve actuation means includes a guide member driven in a cylindrical motion.
9. The apparatus of anyone of claims 2-8, wherein the adjustment means includes a guide element having a path adapted to receive the guide member.
10. The apparatus of claim 9, wherein the guide element is movable from a first position to a second position, the guide member received in the guide path having a trajectory when the adjustment means is in the first position, different to the trajectory of the guide member when the adjustment means is in a second position.

11. The apparatus of anyone of claims 3 to 10, wherein the valve opening and closing angle in relation to crankshaft position is variable between two predetermined angles.
12. The apparatus of claim 11, wherein at least 3 angles are selected over a range of engine speed.
13. An apparatus for adjusting the motion characteristics of a valve, including adjustment means to adjust the valve motion in accordance with the adjustment means travel along a non-straight path.
14. An apparatus as claimed in claim 13, wherein the adjustment means has a guide element and a guide member, wherein the guide member moves along a guide path in the guide element and along a guide path in a valve actuation means, causing the valve actuation means to move.
15. An apparatus as claimed in claim 13, including a rocker arm rotatably mounted around a point, a guide plate adjustable in relation to the point and an actuation means for moving a guide member along a guide path in the rocker arm and the guide plate wherein differences in the guide paths is reduced when the guide plate is in a first position, compared to the difference in the guide path when the guide plate is in a second position.
16. The apparatus of any preceding claim, wherein the motion characteristics of an inlet valve are variable independently of the motion characteristics of the exhaust valve.
17. An adjustment means for use in an apparatus for adjusting the motion characteristics of a valve comprising a plate having a guide path.
18. The apparatus of claim 17 including a valve actuator having a guide path difference from the guide path of the adjustment means.

19. The apparatus of claim 17, wherein the amount of non-alignment of the guide path of the adjustment means and the guide path of the valve actuator determines the motion characteristics of the valve actuator.

20. The apparatus of claim 18, wherein varying the position of the adjustment means changes the alignment between the guide path of the adjustment means and the guide path of the valve actuator thus adjusting the motion characteristics of the valve.

21. An apparatus for adjusting a motion characteristics of a valve including a first guide path and a second guide path wherein the motion characteristics of the valve are determined by differences in shape and/or alignment between the first and second guide paths.

22. The apparatus of any preceding claim, including a desmodromic valve actuation means.

23. An apparatus for adjusting the clearance of a valve actuated by a desmodromic valve actuation means including a valve having a threaded end portion.

24. The apparatus of claim 23, wherein the valve actuation means is located between a lower flanged member and upper flanged member, the upper flanged member being threadedly attached to the threaded valve portion.

25. The apparatus of claim 24, wherein the upper flanged portion and the lower flanged portion are spaced by a shim.

26. The apparatus of any preceding claim, wherein the valve actuation means includes a valve contact surface having a substantially constant radius.

AMENDED CLAIMS

[received by the International Bureau on 15 June 1998 (15.06.98);
original claims 1-26 replaced by new claims 1-23 (4 pages)]

1. An apparatus for adjusting the motion characteristics of a valve over a range of opening and closing angles including a member in communication with a valve actuator, wherein the member travels along a non-straight path in a guide, and the position of the guide is varied to adjust the direction of motion of the member.
2. The apparatus of claim 1 wherein the valve actuator is a pivotally located arm.
3. The apparatus of claims 1 or 2 wherein the guide includes a plate having a non-straight path.
4. The apparatus of any one of claims 1 to 3, wherein at least 3 different opening and closing angles are able to be selected over a range of engine speed.
5. An apparatus for adjusting the motion characteristics of a valve, including adjustment means to adjust the valve motion including an adjustment member and a guide member, the guide member moving along a guide path in the adjustment member and also along a guide path in a valve actuator, wherein the guide path in the adjustment member and the guide path in the valve actuator are not collinear, and the guide member moving along both paths causes the valve actuator to move relative to the adjustment member due to the differences in these paths.
6. An apparatus as claimed in claim 5, wherein the valve actuator includes a rocker arm rotatably mounted around a point, the adjustment member includes a guide plate adjustable in relation to that point, and an actuation means for moving a guide member along a guide path in the rocker arm and a guide path in the guide plate wherein differences in the respective guide paths are reduced

AMENDED SHEET (ARTICLE 19)

when the guide plate is in a first position, compared to the differences in the guide paths when the guide plate is in a second position.

7. An adjustment means for use in an apparatus for adjusting the motion characteristics of a valve comprising a plate having a guide path.

8. The apparatus of claim 7 including a valve actuator having a guide path different from the guide path of the adjustment means.

9. The apparatus of claim 8, wherein the amount of non-alignment of the guide path of the adjustment means and the guide path of the valve actuator determines the motion characteristics of the valve actuator.

10. The apparatus of claim 9, wherein varying the position of the adjustment means changes the alignment between the guide path of the adjustment means and the guide path of the valve actuator thus adjusting the motion characteristics of the valve.

11. An apparatus for adjusting the motion characteristics of a valve including a first guide path and a second guide path wherein the motion characteristics of the valve are determined by differences in shape and/or alignment between the first and second guide paths.

12. The apparatus of claim 11, wherein the first guide path is located in an adjustment member, which moves relative to the valve actuator to alter the differences between the shape and/or alignment of the first and second guide paths.

13. An apparatus for adjusting the motion characteristics of a valve including an adjustment means operatively situated between a valve actuation means and the valve, having an adjustment member with a guide path adapted to receive a guide member, the adjustment member being moveable between a

first position and a second position, wherein the guide member travelling along the guide path of the adjustment member has a different trajectory when the adjustment member is in the first position than when the adjustment member is in the second position.

14. The apparatus of claim 13 wherein the valve actuation means includes the means for determining the base timing of the valve actuation.

15. The apparatus of claim 13 or 14 where the guide member moves cyclicly.

16. The apparatus of any one of claims 13 to 15 where the position of the adjustment member can be varied over a given range of engine speeds.

17. The apparatus of any one of claims 13 to 16 where the position of the adjustment member can be varied over a given range of engine loads.

18. An apparatus for adjusting the motion characteristics of a valve including an adjustment member having a guide path, a guide element having a guide path and a valve actuating member, wherein the adjustment member moves relative to the guide element, thus modifying the motion of the guide member moving along the first and second guide paths which also modifies the motion of the guide element.

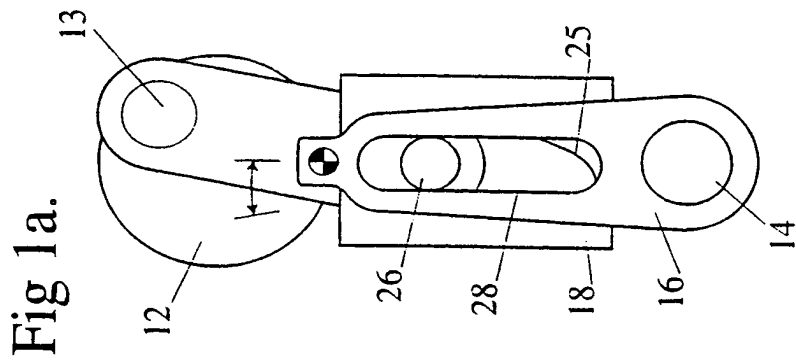
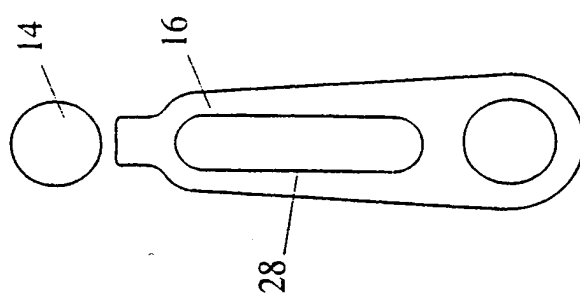
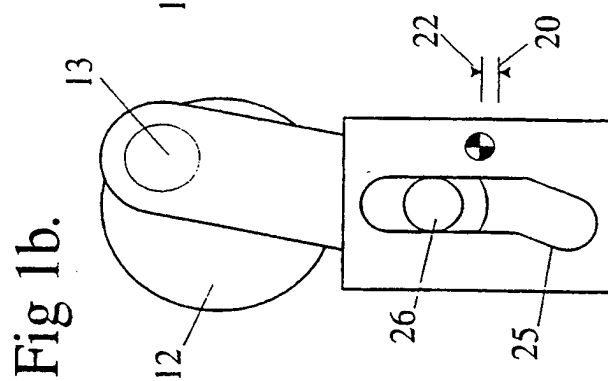
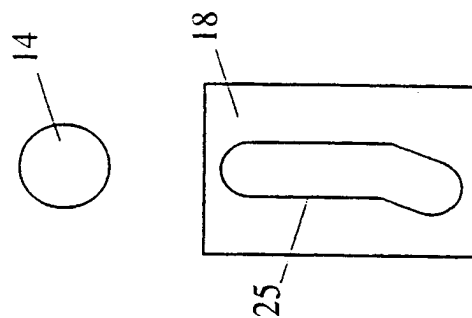
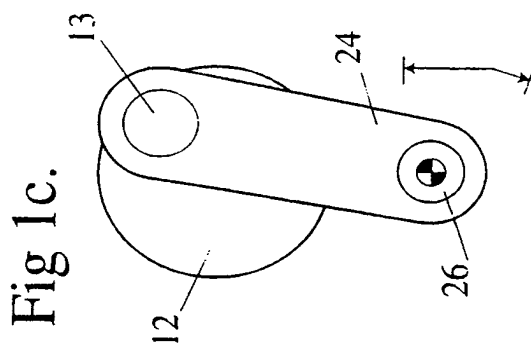
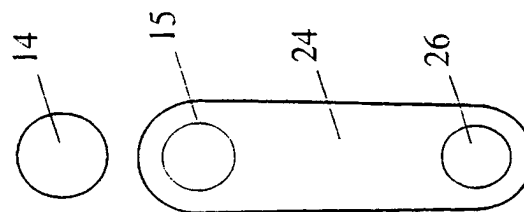
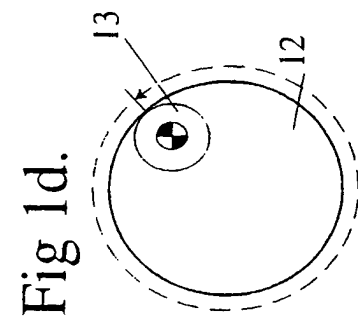
19. The apparatus of claim 18 wherein the adjustment member and guide element are plates.

20. The apparatus of any one of the preceding claims wherein the base timing of the adjustment mechanism is not varied.

21. The apparatus of any of claims 13 to 15 wherein the guide element is pivotally located.

22. The apparatus of any one of the preceding claims wherein the adjustment member is slidably located with respect to the pivot point of the guide element.

23. The apparatus of any one of the preceding claims, wherein the motion characteristics of an inlet valve are variable independently of the motion characteristics of the exhaust valve.



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Fig 2a.

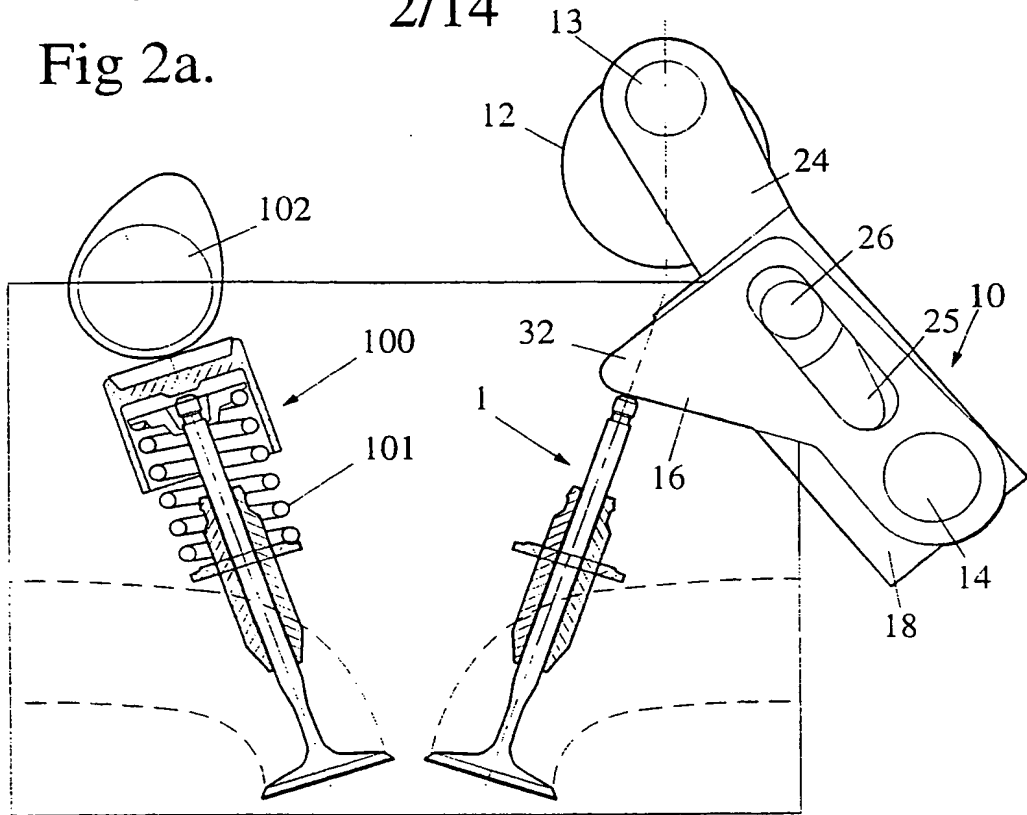
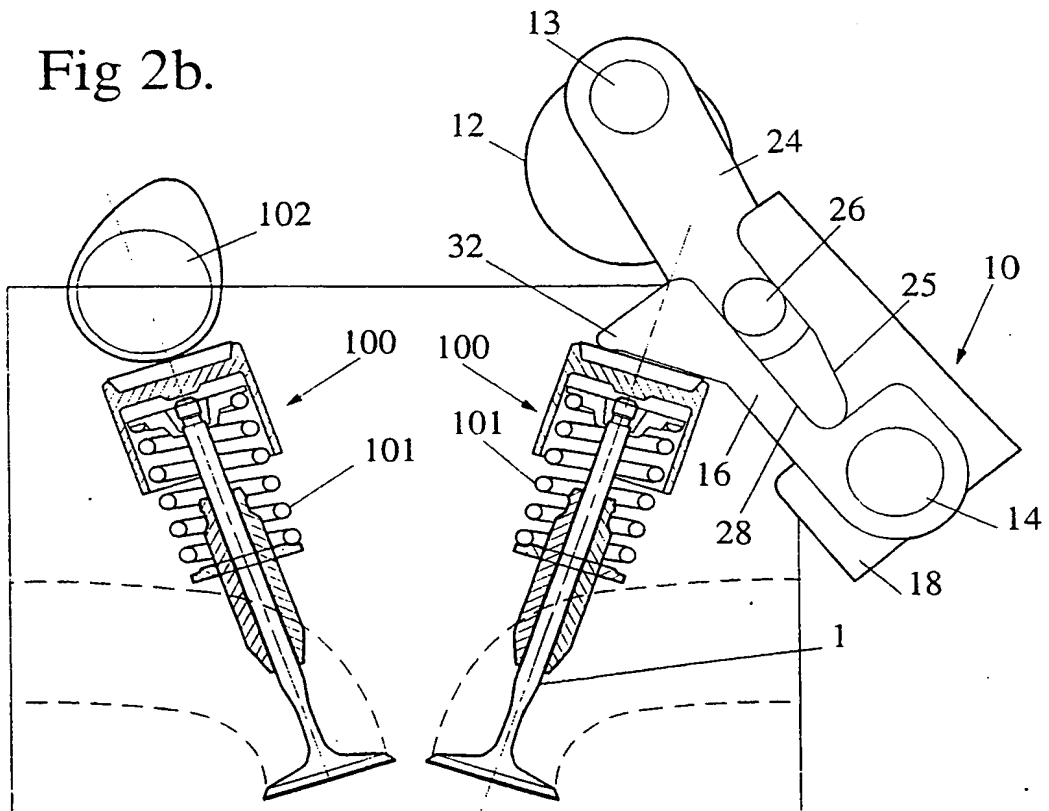


Fig 2b.



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Fig 3.

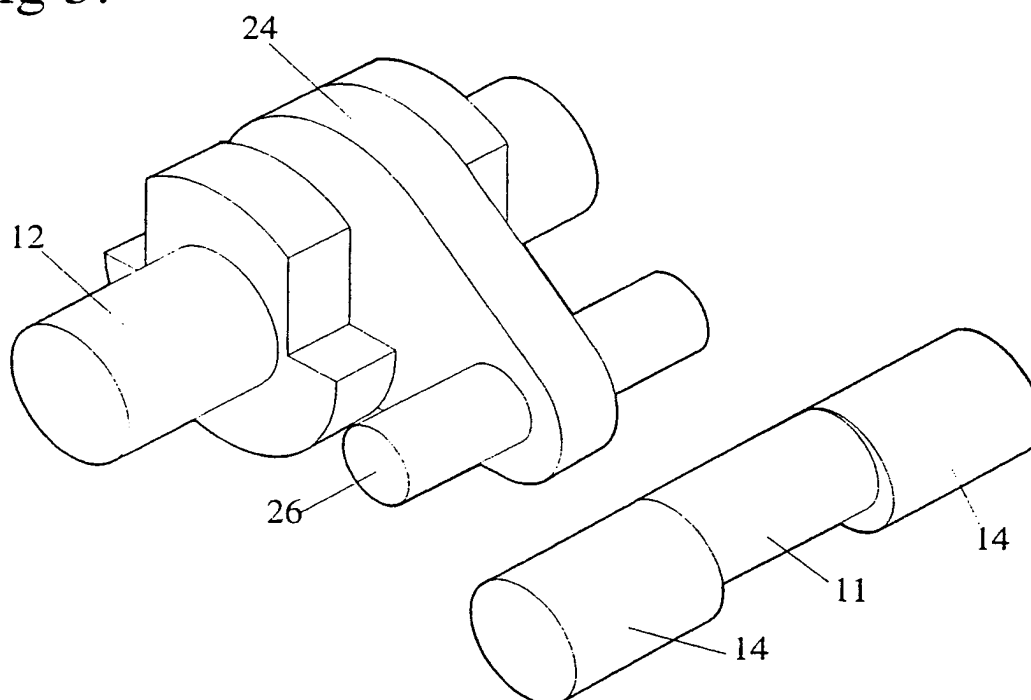
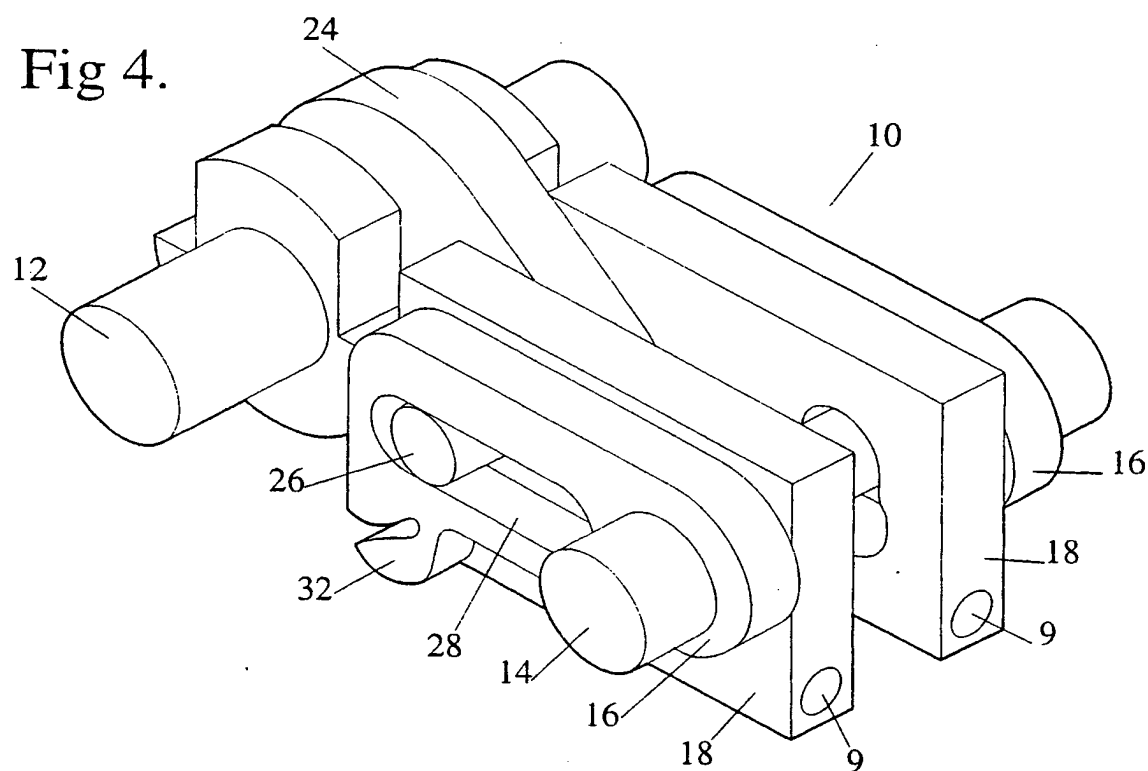


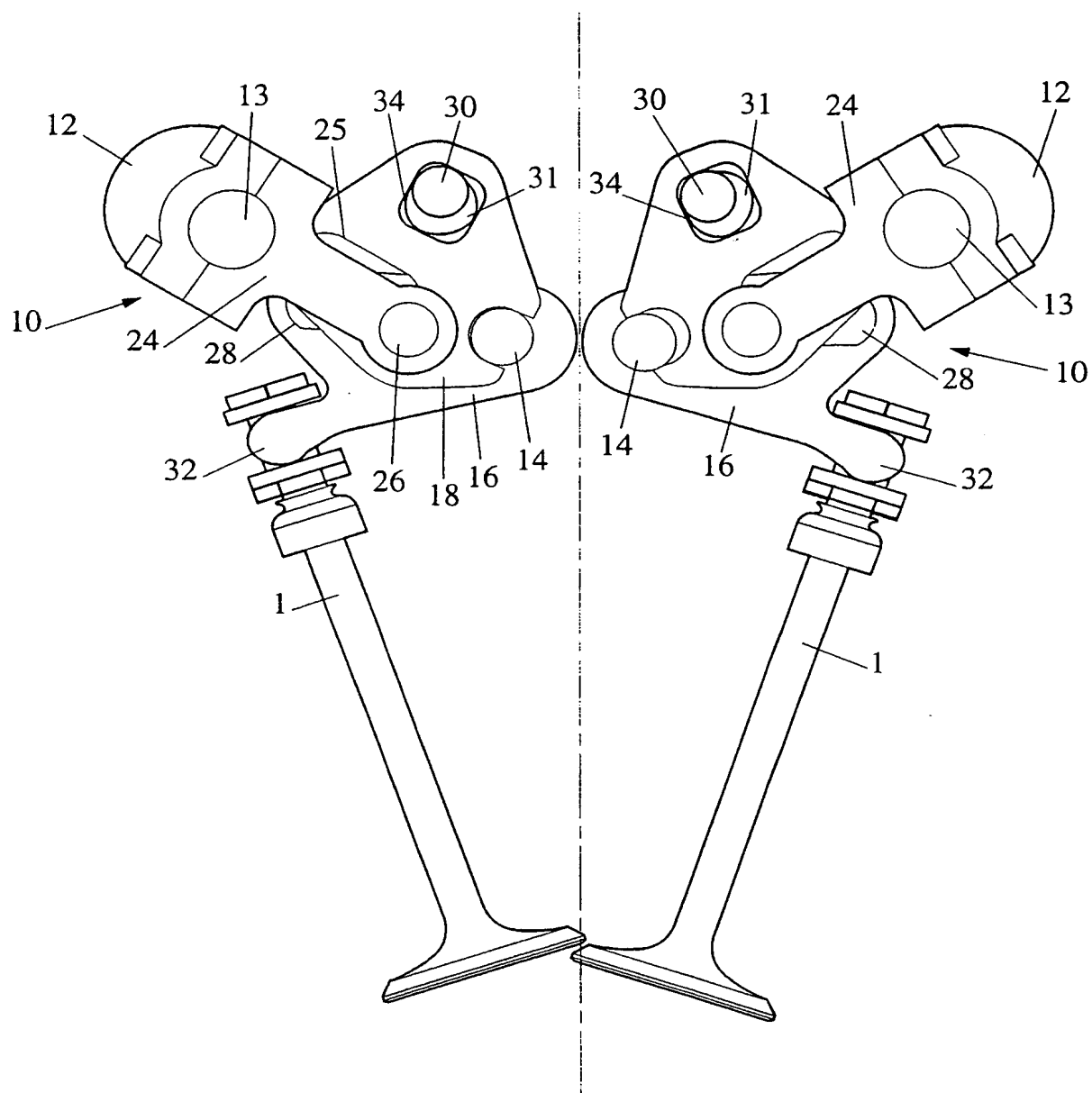
Fig 4.



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Fig5a.

Fig5b.

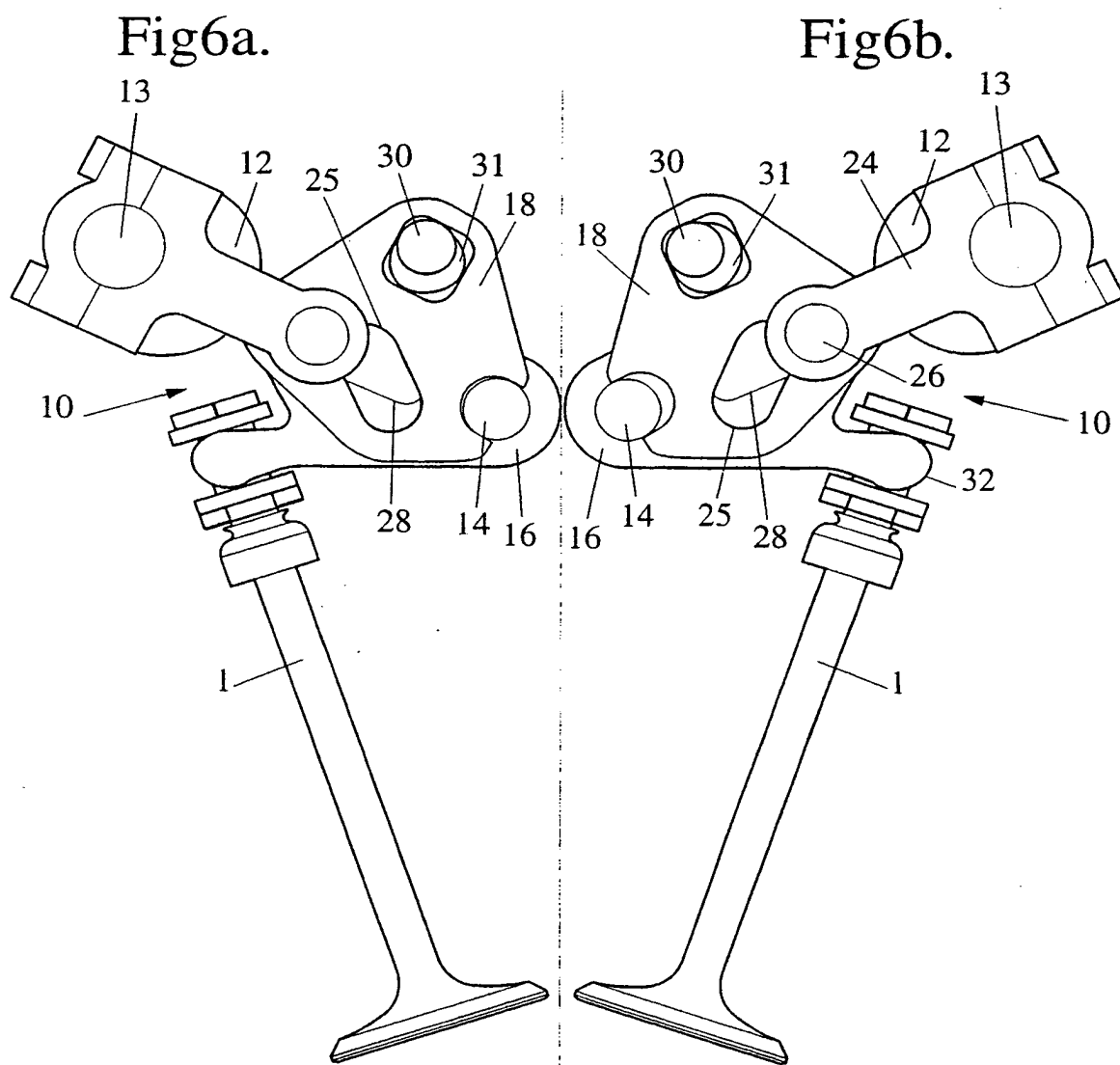


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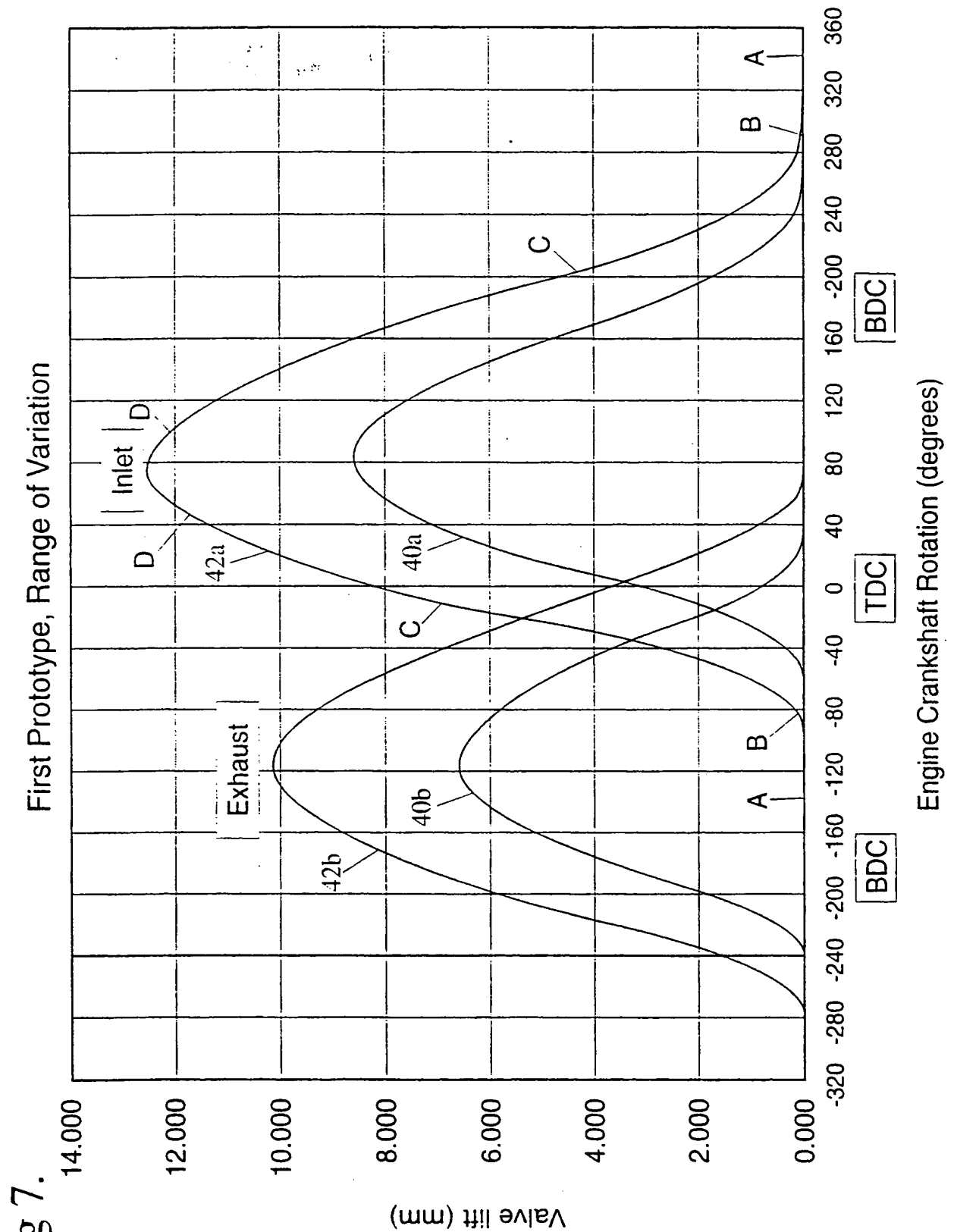
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Fig 8.

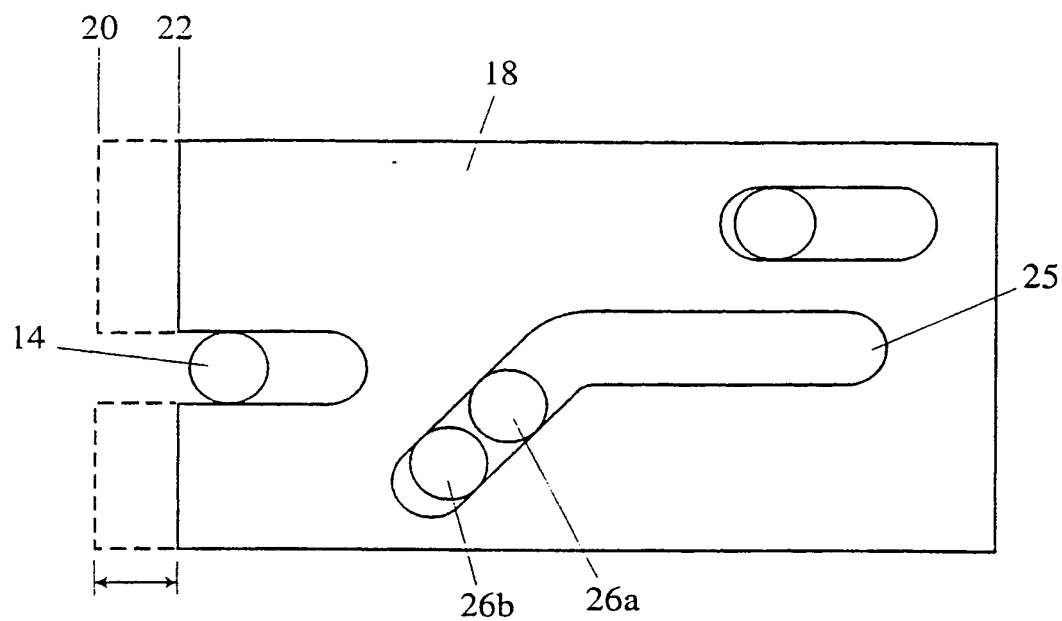
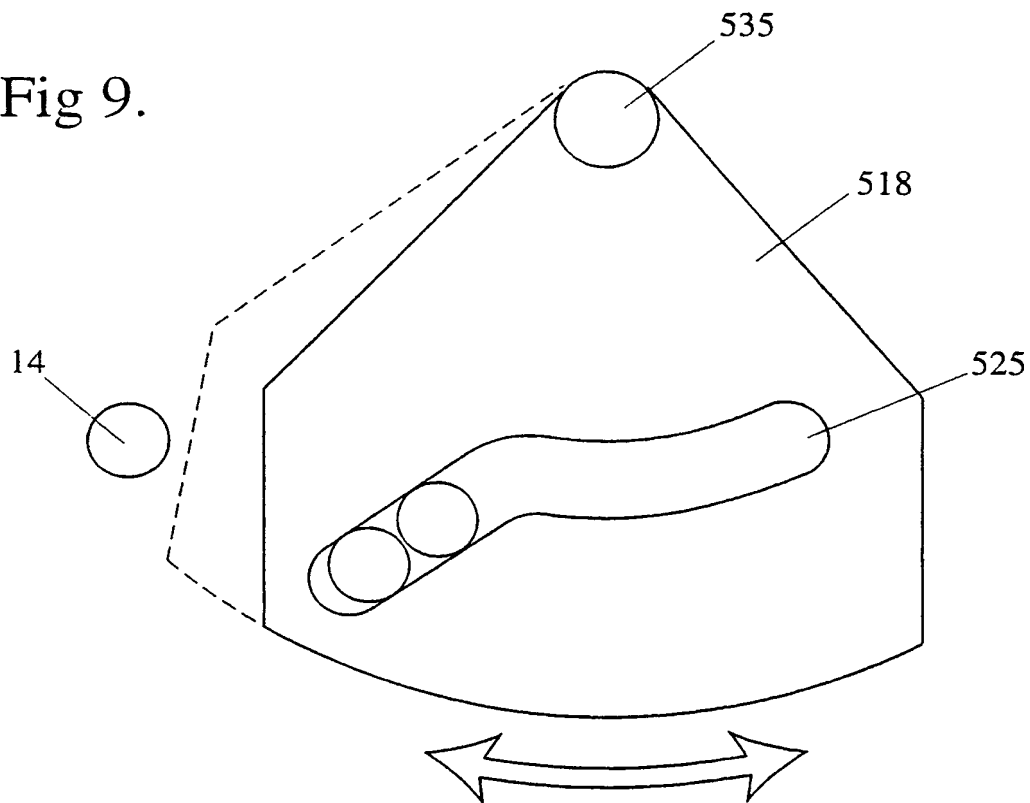


Fig 9.



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Fig 10c.

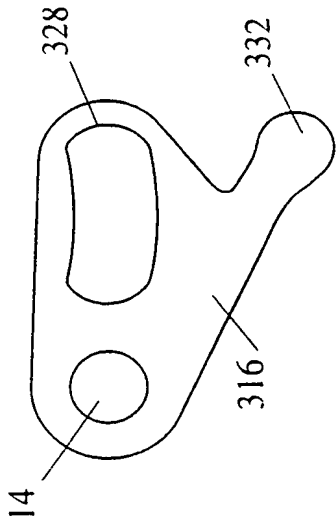


Fig 10d.

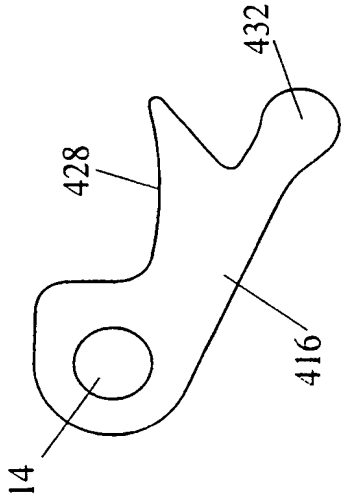


Fig 10a.

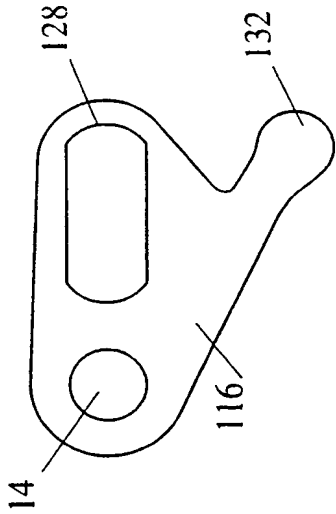
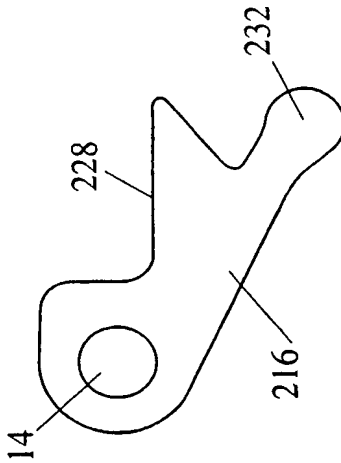


Fig 10b.



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Fig 11.

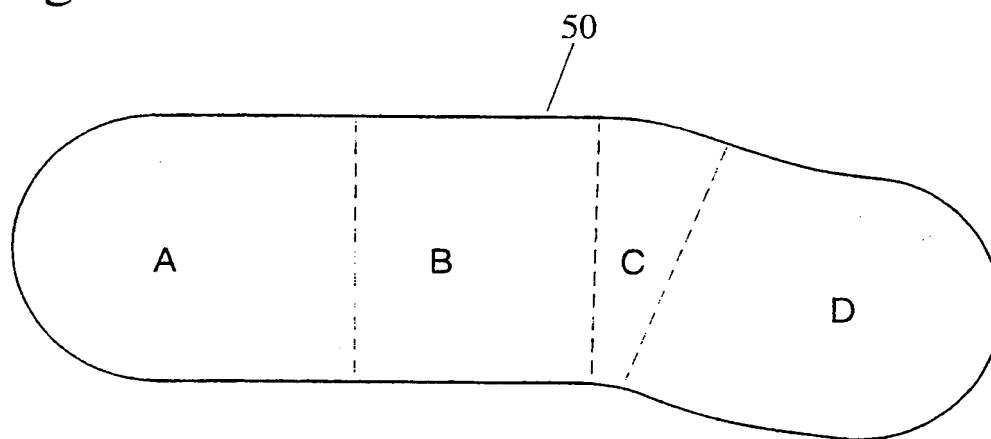
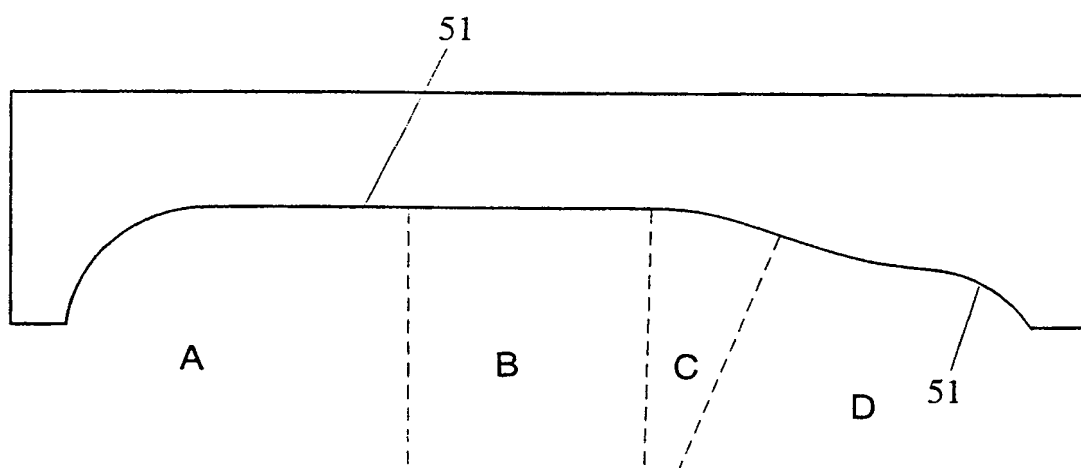


Fig 12.



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Fig 13.

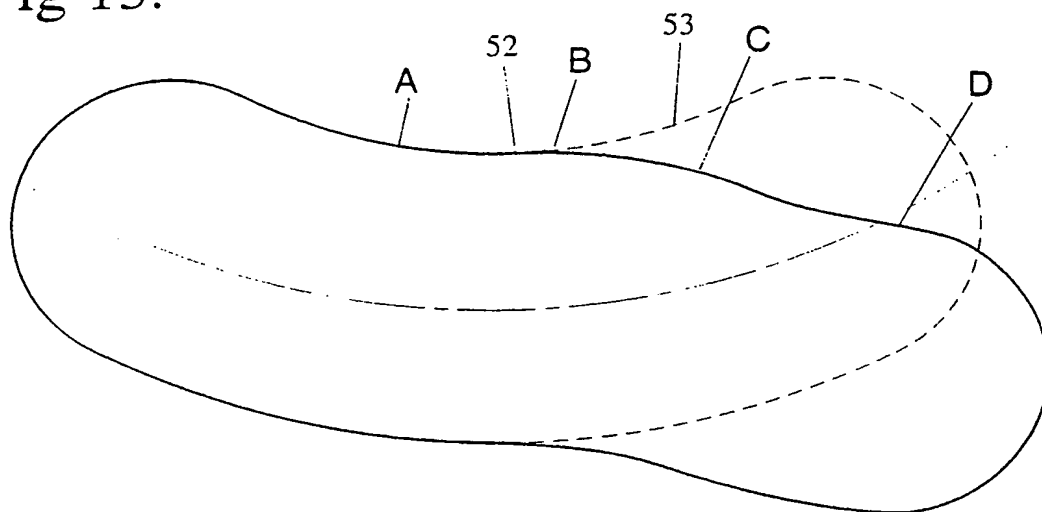
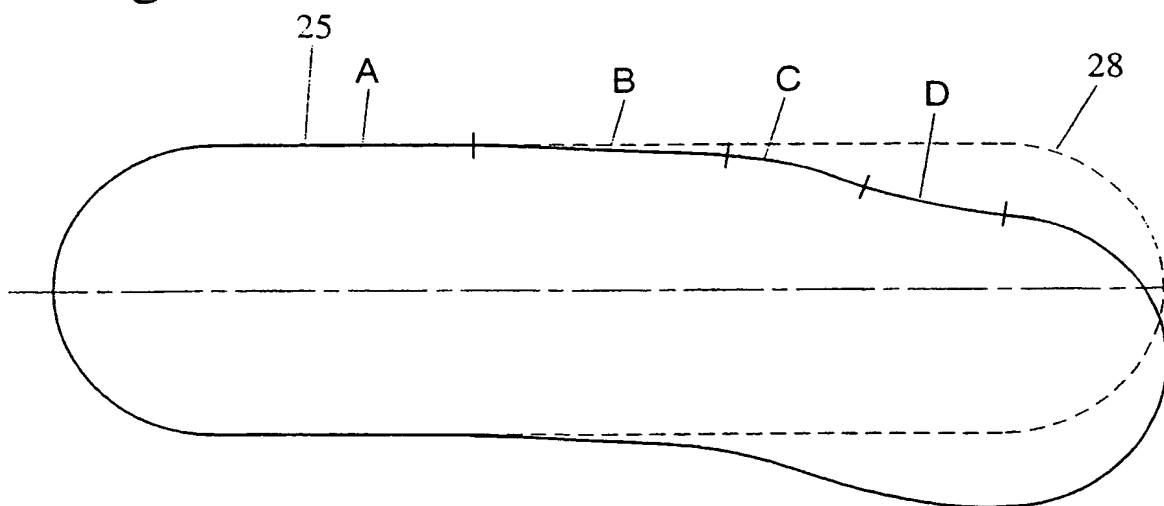
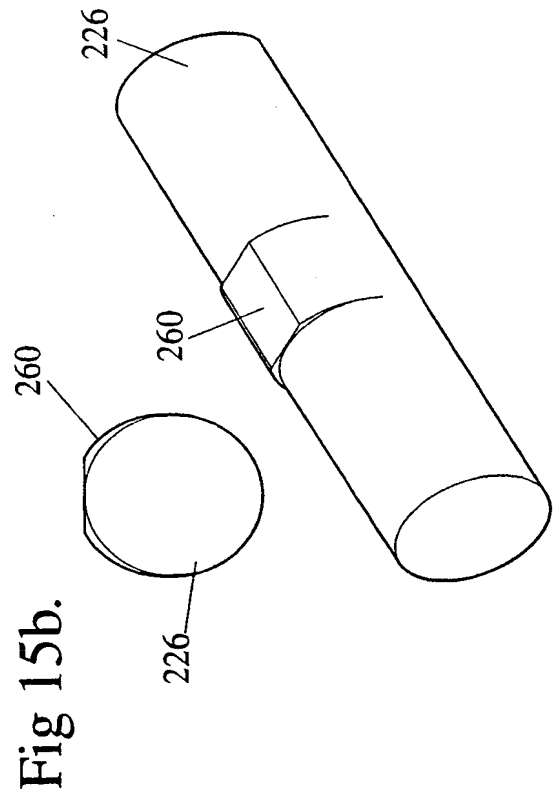
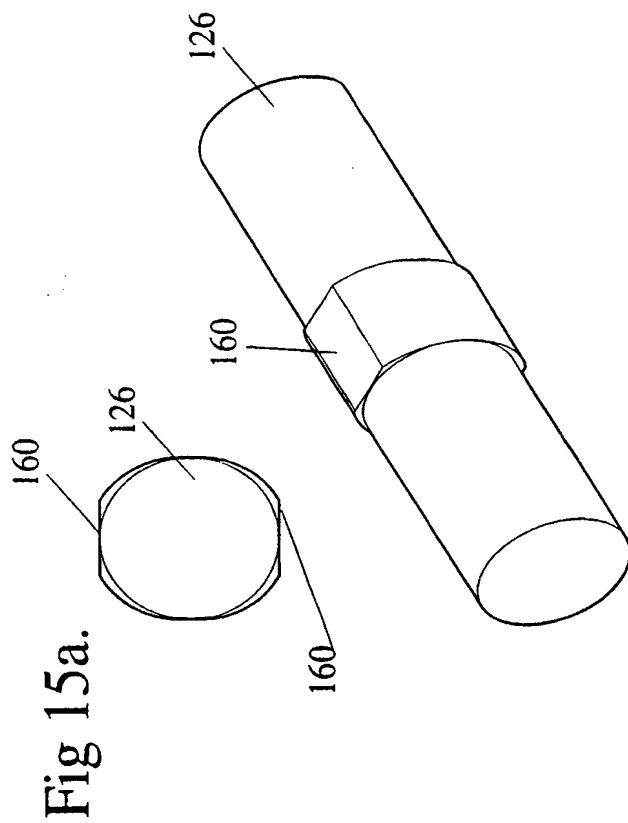
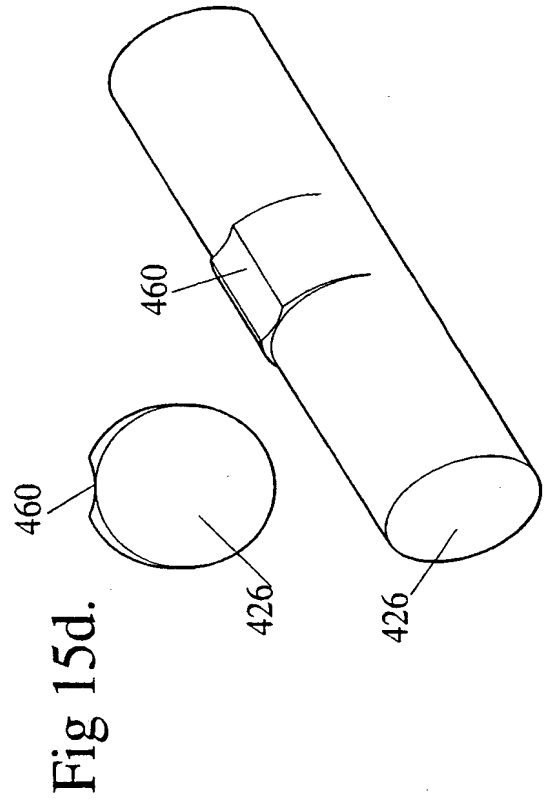
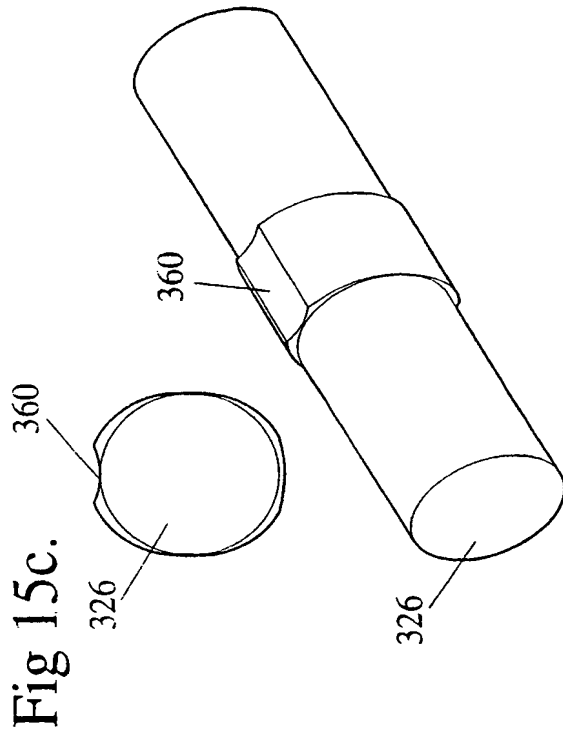


Fig 14.



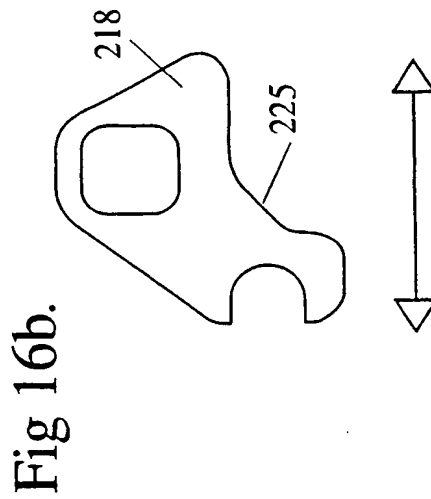
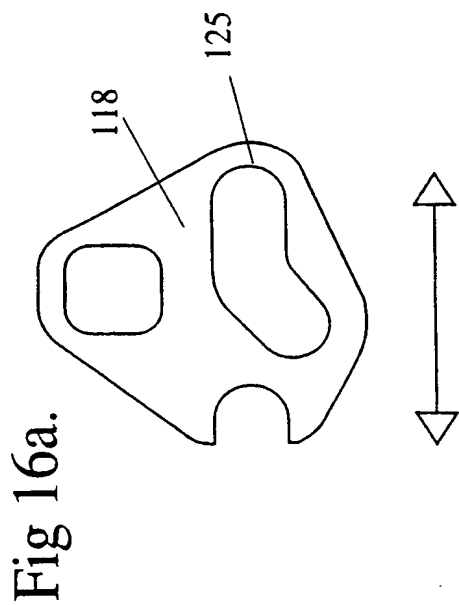
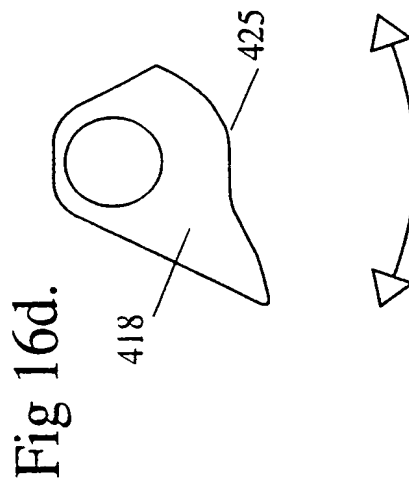
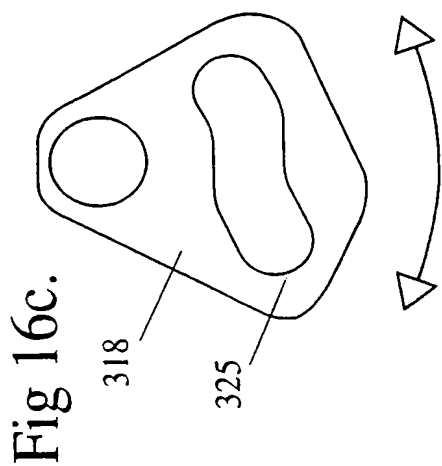
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Fig 17b.

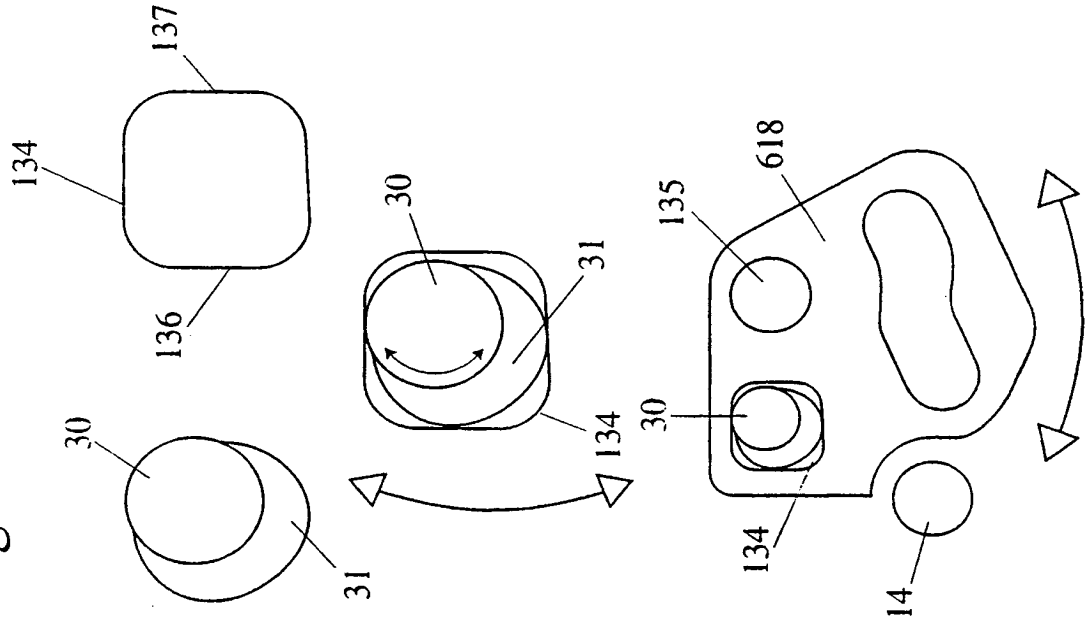
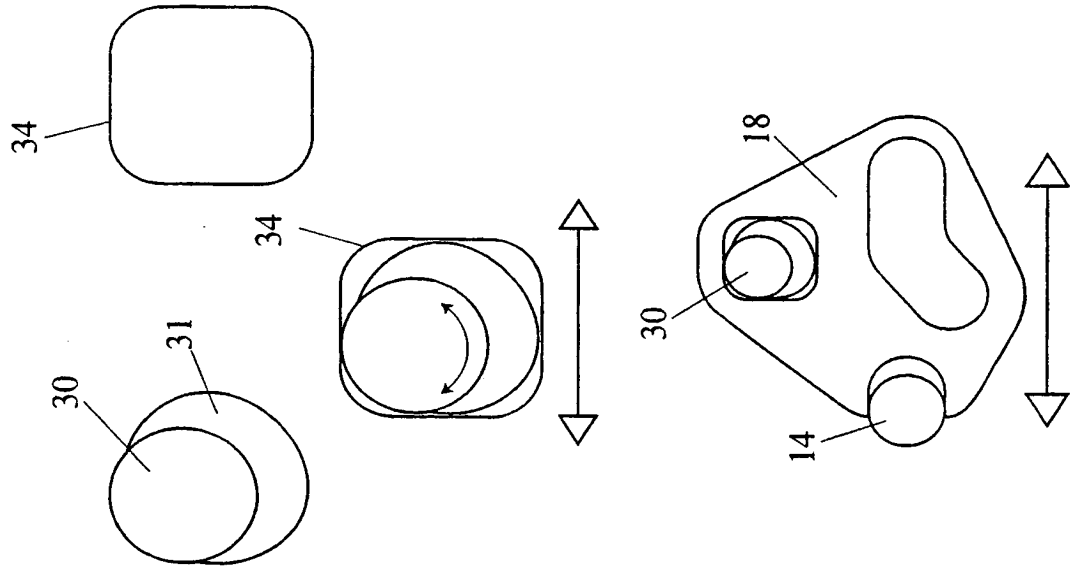


Fig 17a.



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Fig 18a.

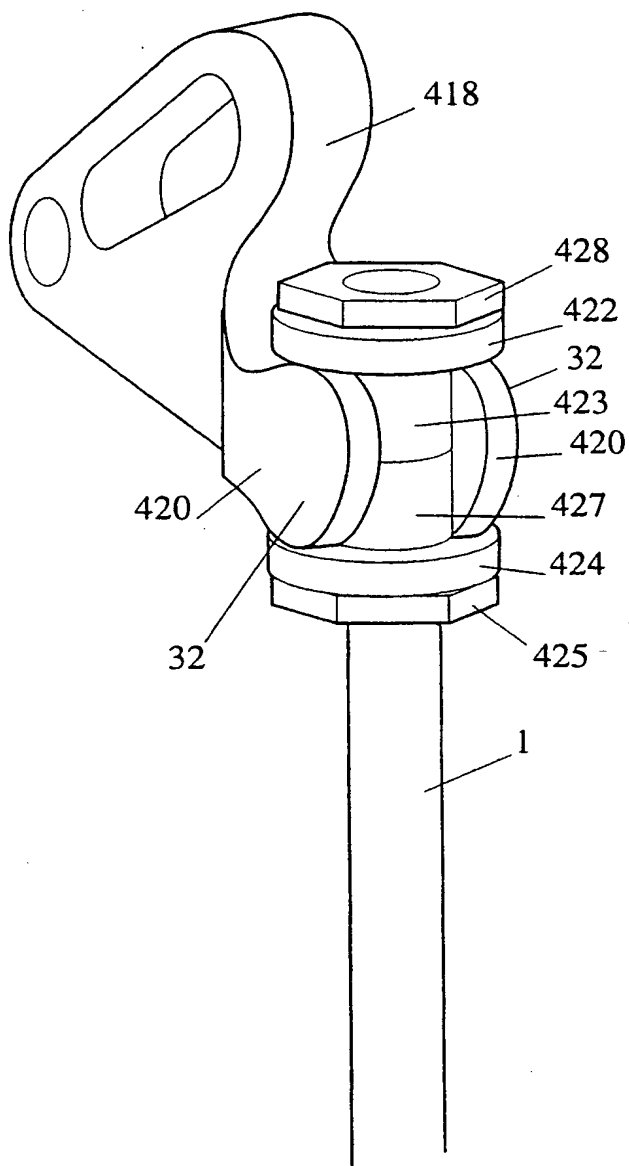
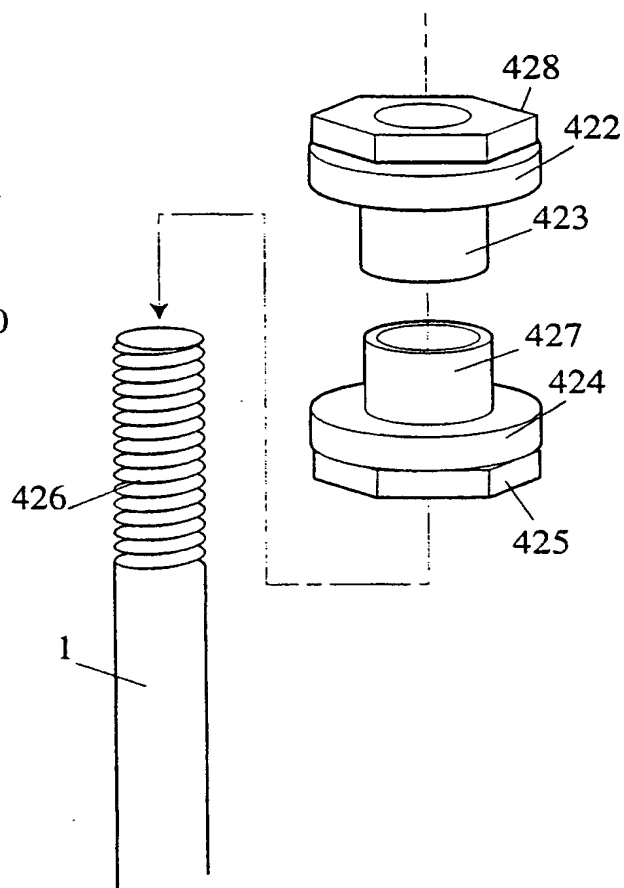


Fig 18b.



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INTERNATIONAL SEARCH REPORT

International Application No.
PCT/AU 98/00090

A. CLASSIFICATION OF SUBJECT MATTER												
Int Cl ⁶ : F01L 1/12, 1/20, 1/30												
According to International Patent Classification (IPC) or to both national classification and IPC												
B. FIELDS SEARCHED												
Minimum documentation searched (classification system followed by classification symbols) F01L 1/12, 1/20, 1/30, F01L: F01B; F02D: F02B and F02M												
Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched AU: IPC as above												
Electronic data base consulted during the international search (name of data base and, where practicable, search terms used) WPAT: (1) (ENGINE# OR MOTOR#) AND VALV: AND ADJUST: AND PLATE AND (SLOT: OR PATH: OR GUID:) (2) (ENGINE# OR MOTOR#) AND VALV: AND (ADJUST: OR ALTER: OR VARY: OR VARIABLE#) AND DESMODROMIC: AND (THREAD: OR SCREW:)												
C. DOCUMENTS CONSIDERED TO BE RELEVANT												
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.										
X	GB 1496513 A (SCARROTT) 30 December 1977 whole document	1-14, 16, 26										
X	US 4364341 A (HOLTMANN) 21 December 1982 column 3 line 48 to column 6 line 17, claims 1 and 2 and Figures 1-5	1-14, 16, 26										
X	EP 311282 A (JAGUAR CARS LIMITED) 12 April 1989 column 3 line 41 to column 5 line 56, claim 10 and Figures 3-8	1-14, 16, 26										
X	GB 343688 A (MORRIS) 26 February 1931 page 2 lines 73 to 32 and Figures 1 and 2	1-14, 16, 26										
<input checked="" type="checkbox"/> Further documents are listed in the continuation of Box C <input checked="" type="checkbox"/> See patent family annex												
<p>* Special categories of cited documents:</p> <table border="0"> <tr> <td>"A" document defining the general state of the art which is not considered to be of particular relevance</td> <td>"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention</td> </tr> <tr> <td>"E" earlier document but published on or after the international filing date</td> <td>"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone</td> </tr> <tr> <td>"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)</td> <td>"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art</td> </tr> <tr> <td>"O" document referring to an oral disclosure, use, exhibition or other means</td> <td>"&" document member of the same patent family</td> </tr> <tr> <td>"P" document published prior to the international filing date but later than the priority date claimed</td> <td></td> </tr> </table>			"A" document defining the general state of the art which is not considered to be of particular relevance	"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention	"E" earlier document but published on or after the international filing date	"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone	"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)	"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art	"O" document referring to an oral disclosure, use, exhibition or other means	"&" document member of the same patent family	"P" document published prior to the international filing date but later than the priority date claimed	
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"O" document referring to an oral disclosure, use, exhibition or other means	"&" document member of the same patent family											
"P" document published prior to the international filing date but later than the priority date claimed												
Date of the actual completion of the international search 25 March 1998		Date of mailing of the international search report 31 MAR 1998										
Name and mailing address of the ISA/AU AUSTRALIAN PATENT OFFICE IP AUSTRALIA PO BOX 200 WODEN ACT 2606 AUSTRALIA Facsimile No.: (02) 6285 3929		Authorized officer LIONEL BOPAGE Telephone No.: (02) 6283 2153										

INTERNATIONAL SEARCH REPORTInternational Application No.
PCT/AU 98/00090

C (Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT		
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	FR 573903 A (BÉNIER) 2 July 1924 whole document, in particular, the figure	22-25
X	US 4495902 A (BURANDT) 29 January 1988 claim 18 and Figure 6	17
X	G D Hiscox, "Gas, Gasoline, and Oil Engines including Producer-gas Plants", published 1906, by the Norman W Henley Publishing Co (New York) page 126, Figure 71	22-25

INTERNATIONAL SEARCH REPORT

International Application No.
PCT/AU 98/00090

Box I Observations where certain claims were found unsearchable (Continuation of item 1 of first sheet)

This International Search Report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

1. ☒ Claims Nos.:
because they relate to subject matter not required to be searched by this Authority, namely:

At least claims 1-9 of the present application are of very wide scope as they fail to define any specific structural features of the apparatus and/or adjusting means. These claims are formulated in a very vague and broad manner. For economic reasons a complete search of these claims is impossible.
Claims incompletely searched: 1-9, 11, 13, 16, 17, 21 and 23.
2. ☒ Claims Nos.:
because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out, specifically:

At least claims 1-9 of the present application are of very wide scope as they fail to define any specific structural features of the apparatus and/or adjusting means. These claims are formulated in a very vague and broad manner. For economic reasons a complete search of these claims is impossible.
Claims incompletely searched: 1-9, 11, 13, 16, 17, 21 and 23.
3. ☐ Claims Nos.:
because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a)

Box II Observations where unity of invention is lacking (Continuation of item 2 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:

The application does not comply with the requirements of unity of invention. Claims 1-22 and claims 23-26 define two different inventions due to reasoning provided in the additional sheet.

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims
2. ☒ As all searchable claims could be searched without effort justifying an additional fee, this Authority did not invite payment of any additional fee.
3. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:
4. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

Remark on Protest

- ☐ The additional search fees were accompanied by the applicant's protest.
☐ No protest accompanied the payment of additional search fees.

INTERNATIONAL SEARCH REPORT

International Application No.
PCT/AU 98/00090

Box II (Continuation of item 2)

The international application does not comply with the requirements of unity of invention because it does not relate to one invention or to a group of inventions so linked as to form a single general inventive concept. In coming to this conclusion the International Searching Authority has found that there are different inventions as follows:

1. Claims 1-22 are directed to an apparatus for adjusting the motion characteristics of a valve, comprising an adjustment means having a plate with a guide path. It is considered that the adjustment means having a plate with a guide path comprises a first "special technical feature".
2. Claims 23-26 are directed to an apparatus for adjusting the clearance of a valve. It is considered that the adjusting of valve clearance comprises a second "special technical feature".

Since the abovementioned groups of claims do not share any of the technical features identified, a "technical relationship" between the inventions, as defined in PCT rule 13.2 does not exist. Accordingly the international application does not relate to one invention or to a single inventive concept, a priori.

However since all these inventions share the same classification under the IPC they could be searched together without effort which would warrant an additional fee.

INTERNATIONAL SEARCH REPORT

International Application No.

This Annex lists the known "A" publication level patent family members relating to the patent documents cited in the above-mentioned international search report. The Australian Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

Patent Document Cited in Search Report			Patent Family Member			
GB	1496513	NONE				
US	4364341	DE	3022188			
EP	0311282	JP	1147106	US	4898130	
US	4495902	EP	125096	JP	59208110	US 4723515